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An open steam power cycle used for IC engine exhaust gas energy recovery

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ABSTRACT

In order to improve IC engine energy utilization efficiency, an open steam power cycle used for IC engine exhaust gas energy recovery is proposed. The bottom cycle concept is designed on a four-cylinder naturally aspirated IC engine: with three cylinders taken as ignition cylinder, the last one is used for steam expansion cylinder; IC engine exhaust pipe is coupled with a Rankine steam cycle system which uses the high temperature exhaust gas to generate steam; then, the steam is injected into steam expansion cylinder and expands in the cylinder. In this way, the Otto cycle (or diesel cycle) of traditional IC engine and the steam expansion cycle (open Rankine cycle) are coupled on IC engine. On this basis, the energy recovery potential of this bottom cycle is studied by cycle processes calculation and parameters analysis. The research results show that the recovery efficiency of exhaust gas energy is mainly limited by exhaust gas temperature. The maximum bottom cycle power can reach 19.2 kW and IC engine thermal efficiency can be improved by 6.3% at 6000 r/min. All those can prove this novel bottom cycle concept has larger potential for energy saving and emission reduction on IC engine.

1. Introduction

Energy saving and emission reduction have become the development impetus and goal of modern internal combustion engine (IC engine) [1,2]. In all kinds of technologies for energy saving and emission reduction on IC engine, including GDI [3] and HCCI [4], waste heat recovery has the largest energy saving potential as it can be demonstrated from analyzing the IC engine heat balance [5,6]. Compared to cooling water energy, exhaust gas energy has higher recovery potential because exhaust gas has higher temperature and contains more exergy [7]. Thus, exhaust gas energy recovery is one of the most important ways to improve IC engine thermal efficiency and to reduce fuel consumption. Many scientists and engineers have done lots of research on it [2,7–14], and many concepts have been proposed for exhaust gas energy recovery, such as compound turbocharged, combined heat and power generation and exhaust thermal different power plant, etc. For example, He et al. [7] have designed a combined thermodynamic cycle used for waste heat recovery on IC engine; Hung et al. [8] have studied the organic working fluids on system efficiency of an ORC using low grade energy sources; Hsiao et al. [14] have proposed a mathematic model of thermoelectric module with applications on waste heat recovery from automobile engine, etc. However, there are lots of problems in the existing recovery methods and concepts. Usually, those methods based on exhaust gas direct expansion not only correspond to relatively low energy recovery efficiencies, but also influence the working performance of IC engine through bringing additional pumping losses during the exhaust gas process; those methods based on closed thermodynamic cycle require additional thermal equipment to add to IC engine, which results in the consequence that bottom cycle system is very complex and the cost is very high, etc. Based on the above considerations, in this paper, the authors propose a kind of novel approach to recover IC engine exhaust gas energy by using open steam power cycle. After the bottom cycle model for exhaust gas energy recovery is presented, the effects of bottom cycle parameters on exhaust gas energy recovery efficiency, IC engine energy saving potential and bottom cycle performances are studied. All those can demonstrate this bottom cycle concept for exhaust gas energy recovery has larger energy saving potential and good application prospect on IC engine.

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2. IC engine exhaust gas energy recovery model

2.1. Principles of bottom cycle model

The bottom cycle model for recovering IC engine exhaust gas energy is proposed as shown in Fig. 1, and the working principles are described as follows. On a naturally aspirated four-cylinder IC engine, three cylinders are taken as ignition cylinders, whose working principles are the same as that of traditional four strokes IC engine, while the last one is taken as steam expansion cylinder. That is to say, this cylinder is only used for steam expansion, without fuel injection or combustion. A set of steam power cycle equipment is coupled on IC engine exhaust gas system and cooling system, and it uses IC engine exhaust gas and coolant for high temperature heat source to heat working medium water. Firstly, working medium water is preheated by the IC engine coolant. Secondly, it is boosted to a certain pressure in the pump. Then, it flows into the boiler, in which working medium water is heated into saturated steam by exhaust gas. Next, the saturated steam is heated into superheated steam in the superheater. After that, high temperature steam is injected into the steam expansion cylinder and expands in the cylinder. Finally, the steam after expansion is discharged to surroundings from cylinder exhaust valve. Obviously, this waste heat recovery approach mainly includes two parts: external bottom cycle (open steam Rankine cycle) and steam expansion cycle in expansion cylinder.

As mentioned above, this bottom cycle operates on the principle of open Rankine steam cycle which takes water as working medium. Fig. 2 depicts the state points and thermodynamics processes of working medium corresponding to the bottom cycle. Combined with Fig. 2, the bottom cycle working processes are described in thermodynamic cycle. Process 1-2(2′): working medium water is boosted to a certain pressure in the pump (the pressure can be optimized by calculation). In the T–S diagram of Fig. 2, process 1-2 represents the real compression process while process 1-2′ is the isentropic compression process. Process 2-3: working medium water is heated into saturated steam in the boiler. Process 3-4: saturated steam is heated again by the exhaust gas in the superheater and then it turns into superheated steam, and this process can be regarded as isobaric process. Process 4-5(5′): superheated steam is injected into the expansion cylinder and then it expands in this cylinder. In the T–S diagram, process 4-5 represents the real expansion process while process 4-5′ is the isentropic expansion process with the irreversible loss in the expansion process ignored. Process 5-1: steam after expansion is directly discharged to surrounding through exhaust valve. As the cycle is not closed, it is a kind of open Rankine steam cycle. Thus, in the T–S diagram the process 5-1 is illustrated in dotted line, as shown in Fig. 2. Compared to other closed thermodynamic cycle, this bottom cycle is easier to realize since the cycle processes are simpler. Meanwhile, the bottom cycle effective work directly outputs from IC engine crankshaft and can be used to drive the automobile.

In this bottom cycle for exhaust gas energy recovery, since the steam expansion cylinder is only used for steam expansion, the igniter (or fuel injector) in cylinder head can be removed. As a result, the valve diameters can be enlarged. Moreover, the steam expansion cylinder can be designed to two strokes (that is, steam expansion cylinder only works the expansion stroke and exhaust stroke). By this means, steam expansion cylinder does not experience compression stroke and thus it does not consume compression work. In addition, it can expand twice in each engine cycle (720°crank angle) and output more effective work. For this purpose, the valve system (including the cam, camshaft and valve timing, etc.) should be modified. Firstly, the camshaft speed should be the same as crankshaft speed (that is, camshaft is directly driven by the crankshaft with the speed ratio of 1:1 rather than 1:2). Then, the cam profiles and timing should be also changed. The exhaust valve should always open when the piston works exhaust gas stock from the bottom dead center (BDC) to the top dead center (TDC), while the opening duration of intake valve depends on the flow rate of injected steam. Actually, the opening duration of intake valve is much smaller than that of exhaust valve. The valve timing can be seen from the Fig. 3(a) in the next section. Table 1 gives the cylinder working orders of this bottom cycle concept. As shown in the table, cylinder 1, cylinder 2 and cylinder 3 work in four stocks, while cylinder 4 works in two stocks. Actually, this bottom cycle concept can be also applied on two stocks engine, such as marine engine. Under that circumstance, all the cylinders, including both the firing cylinder and steam expansion cylinder, work in two stocks. What's more, it is easier to modify the valve system on two stocks engine than on four stocks engine.

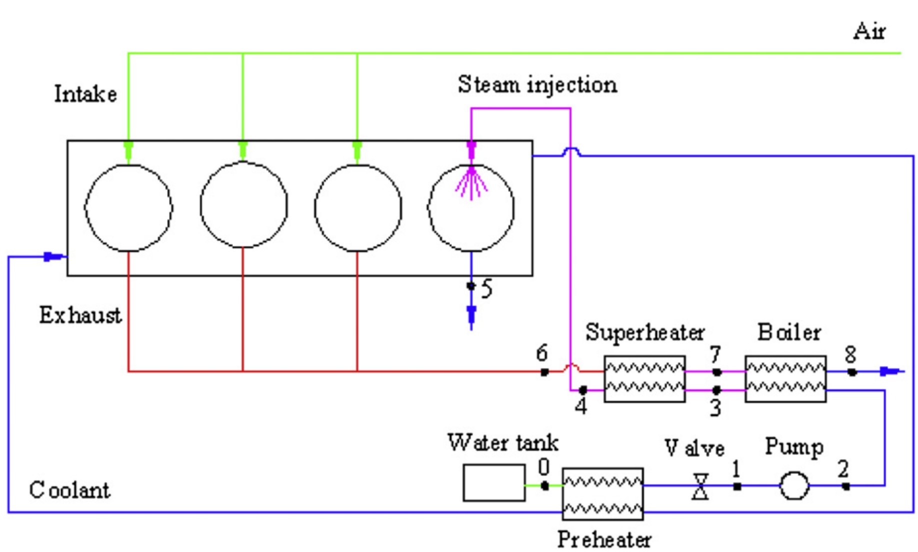

Fig. 1. Conceptual sketch of steam power cycle for IC engine exhaust gas energy recovery.

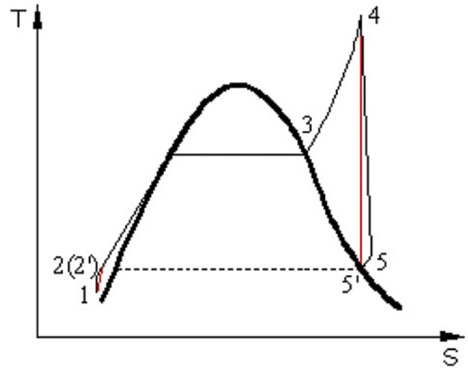

Fig. 2. T–S diagram corresponding to the steam power cycle.

2.2. *Steam expansion process and bottom cycle characteristic analysis*

It is well known that the P–V indicator diagram is one of the best implements to evaluate the capacity to do work in cylinder work processes. The P–V indicator diagram of steam expansion cycle is shown in Fig. 3(a) below. Actually, this figure also shows the valve timing of the steam expansion cylinder. Firstly, the events are described. V0 (piston is at the TDC): exhaust valve closes and intake valve opens (injection starts); V1: intake valve closes (injection finishes); V2 (piston is at the BDC): exhaust valve opens (exhaust starts). Then, the cycle processes are introduced: V0 ~ V1 represents the intake process (steam is injected into cylinder); V1 ~ V2 represents the steam expansion process, which corresponds to the process 4-5(5′) in the T–S diagram of Fig. 2; V2 ~ V2′ ~ V0 is the exhaust process.

Next, the P–V indicator diagram of steam expansion cylinder is compared with that of firing cylinder. For the sake of contrastive analysis, the in-cylinder P–V indicator diagram of traditional four strokes firing cylinder is given in Fig. 3(b). It shows that traditional four strokes firing cylinder consumes lots of compression negative work in the compression stroke, which causes the area of indicated mean effective pressure (IMEP) declines greatly. On the other hand, the exhaust gas pressure is very high since the exhaust gas pipe is connected to complex post-processing system. As a result, the exhaust gas process also consumes some loss work which is called pumping loss (or pumping mean effective pressure, PMEP). Eventually, the net mean effective pressure (NMEP) is relatively small as it equals IMEP minus PMEP. However, things are improved in the steam expansion cylinder. Because the steam expansion cylinder does not require post-processing system, the steam after expansion can be directly discharged to surrounding. As a result, the exhaust back pressure of steam expansion cylinder is smaller than that of firing cylinder. Compared to the firing cylinder, one of the biggest advantages of steam expansion cylinder is that this cylinder is only used for steam expanding without compression stroke, thus it does not consume compression negative work. As mentioned above, because steam expansion cylinder works as the principle of two strokes IC engine, the area surrounded by P-V indicator diagram is the indicated output work of steam expansion cycle. When the highest gas pressure values in the two kinds of cylinders are the same, the P-V indicator diagram of steam expansion cylinder will be more "full" than that of firing cylinder, as shown in Fig. 3(a) and Fig. 3(b) below.

This exhaust gas energy recovery approach realizes not only the combining of traditional IC engine's Otto-cycle (or diesel cycle) and steam power cycle (Rankine cycle), but also the coupling of four strokes working cycle (firing cylinder) and two strokes bottom cycle (steam expansion cylinder) on the same IC engine. Consequently, it integrates the advantages of both the two cycles. As the steam expansion work can directly output from IC engine crankshaft, the transmission efficiency is higher than other waste heat recovery methods. By this means, it can improve IC engine dynamic performance and fuel economy. Also, it can reduce exhaust gas emissions and exhaust pipes thermal stress. Meanwhile, the characteristics of steam power cycle can be summarized as the following aspects. Firstly, because the compressibility of liquid water is very low, the water can be boosted to a required pressure under the premise of consuming little compression work (usually, the compression work consumed by the pump is lower than 2% of bottom cycle output work [15]). Secondly, the liquid water turns into high temperature and high pressure steam which has the ability to do work through phase transition heat transfer. Thirdly, compared to the exhaust gas energy direct recovery method, such as turbo-compound and additional expansion, steam expansion cycle decouples the IC engine exhaust gas pressure and bottom cycle working pressure by the means of heat transfer. As a result, exhaust gas energy can be transferred into bottom cycle working medium on the condition of hardly increasing IC engine exhaust gas back pressure. Last but not the least, since this waste heat recovery method uses water as working medium, bottom cycle system is more lowcost and more environmentally friendly than other recovery means using organic working medium [16,17].

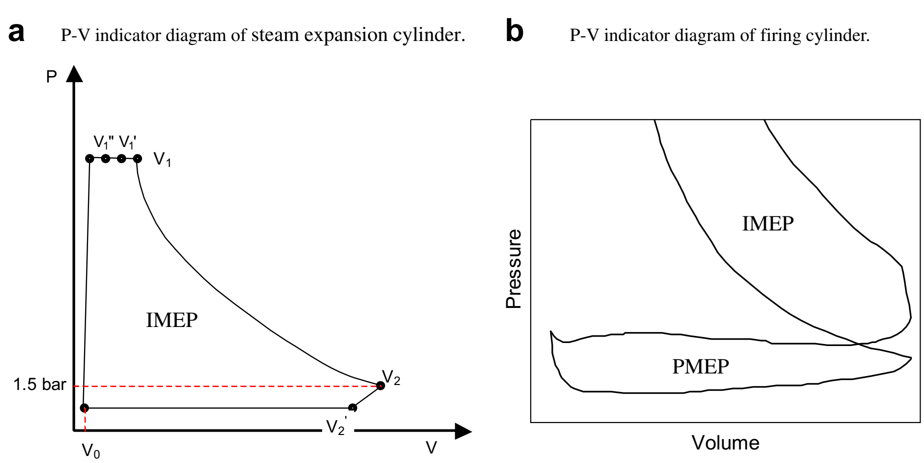

Fig. 3. P–V indicator diagrams of two kinds of cylinder.

Table 1
The cylinder working orders of this bottom cycle.

	Stroke 1	Stroke 2	Stroke 3	Stroke 4
Cylinder 1	Expansion	Exhaust	Intake	Compression
Cylinder 2	Exhaust	Intake	Compression	Expansion
Cylinder 3	Compression	Expansion	Exhaust	Intake
Cylinder 4	Intake (Expansion)	Exhaust	Intake (Expansion)	Exhaust

2.3. Calculation formulas and parameters definition

The exhaust gas energy recovery method proposed in this paper is based on the principle of Rankine steam cycle. Combined with the water and steam T–S diagram shown in Fig. 2, the main performance parameters and calculation formulas of this bottom cycle system are defined. Firstly, the calculation formula for the compression work consumed by pump is given as

$$P_{\mathrm{pum}} = \dot{m}_{\mathrm{st}} \cdot (h_{2'} - h_1)/\eta_{\mathrm{pum}} \tag{1}$$

where, P_{pum} represents the compression work consumed by pump; $\dot{m}_{\mathrm{st}}$ is the mass flow rate of steam generated in the bottom cycle; η_{pum} is the isentropic efficiency of pump; h_1 and $h_{2'}$ are the specific enthalpy of water at the corresponding state points shown in the T–S diagram of Fig. 2. Besides, the specific enthalpy at each state point can be obtained by looking-up the T–S diagram and H–S diagram or by programming calculation on computer according to the cycle process parameters (such as the initial pressure and temperature, etc.) and cycle process features (such as isobaric process and isentropic process, etc.).

The definition of steam expansion work is given as

$$P_{\mathrm{cyl}} = \dot{m}_{\mathrm{st}} \cdot (h_4 - h_{5'}) \cdot \eta_{\mathrm{cyl}} \tag{2}$$

where, P_{cyl} represents the steam expansion work of this bottom cycle; η_{cyl} is the isentropic efficiency of steam expansion process; h_4 and $h_{5'}$ are the specific enthalpy of working medium at the state point 4 and state point 5′, respectively.

Then, the calculation formula for bottom cycle thermal efficiency is given as [15]

$$\eta_{\mathrm{bot}} = \frac{P_{\mathrm{cyl}} - P_{\mathrm{pum}}}{q_h} = \frac{(h_4 - h_{5'}) \cdot \eta_{\mathrm{cyl}} - (h_{2'} - h_1)/\eta_{\mathrm{pum}}}{h_4 - h_2} \tag{3}$$

where, η_{bot} is the bottom cycle thermal efficiency; q_h is the heat absorbed by per unit mass steam in each cycle; h_2 is the specific enthalpy of working medium water at state point 2.

As the specific heat of IC engine exhaust gas monotonically varies with exhaust gas temperature, the energy flow rate of IC engine exhaust gas can be calculated according to the following formula [18]:

$$P_{\mathrm{ex}} = \dot{m}_{\mathrm{ex}} \cdot \int_{T_0}^{T_{\mathrm{ex}}} c_{P\_\mathrm{ex}} \mathrm{d}T \tag{4}$$

where, P_{ex} represents the energy flow rate of IC engine exhaust gas; $\dot{m}_{\mathrm{ex}}$ is the mass flow rate of IC engine exhaust gas; c_{P_ex} is the specific heat of IC engine exhaust gas; T_{ex} is the temperature of IC engine exhaust gas at superheater inlet; T_0 is the temperature of ambient air.

In the heat exchangers (including the boiler and superheater), liquid water is heated by the exhaust gas heat and then turns into superheated steam. The steam state changes from point 2 to point 4 corresponding to the T–S diagram of Fig. 2. At the same time, IC engine exhaust gas is cooled from state point 6 to state point 8. According to the energy conservation law, the following formulas are given:

$$\dot{\Phi}_{6-8} = \dot{m}_{\mathrm{ex}} \cdot (h_6 - h_8) = \dot{\Phi}_{2-4} \tag{5}$$

$$\dot{\Phi}_{2-4} = \dot{m}_{\mathrm{st}} \cdot (h_4 - h_2) \tag{6}$$

where, $\dot{\Phi}_{2-4}$ is the heat flux flows into bottom cycle working medium; $\dot{\Phi}_{6-8}$ is the heat flux flows out of IC engine exhaust gas; h_6 and h_8 are the specific enthalpy of IC engine exhaust gas at state point 6 and state point 8, respectively, as shown in the conceptual sketch of steam power cycle in Fig. 1.

The recovery efficiency of IC engine exhaust gas energy is defined as follows:

$$\eta_{\mathrm{re}} = \frac{P_{\mathrm{bot}}}{P_{\mathrm{ex}}} \cdot 100\% = \frac{P_{\mathrm{cyl}} - P_{\mathrm{pum}}}{P_{ex}} \cdot 100\% \tag{7}$$

where, η_{re} represents the recovery efficiency of IC engine exhaust gas energy; P_{bot} is the output power of this bottom cycle, which equals the steam expansion power minus the power consumed by pump.

Finally, the improvement rate of IC engine thermal efficiency is defined as

$$\eta_{\mathrm{im}} = \frac{P_{\mathrm{bot}}}{\dot{m}_{\mathrm{fue}} \cdot H_u} \cdot 100\% \tag{8}$$

where, η_{im} represents the improvement rate of IC engine thermal efficiency; $\dot{m}_{\mathrm{fue}}$ is the mass flow rate of IC engine fuel consumption; H_u is the low heat value of fuel.

3. Bottom cycle process calculation and parameters setting up

3.1. Boundary conditions for calculation

The research object in this paper is a naturally aspirated gasoline engine, whose name is RAV4. This engine is made by Toyota and it is used for sport utility vehicle. The basic parameters of the RAV4 engine are listed in Table 2. Because the total displacement of this IC engine is 1.998 L, the displacement of each cylinder is 0.5 L. Firstly, this bottom cycle concept for exhaust gas energy recovery was applied on IC engine common operating condition point (speed = 3000 r/min, BMEP = 0.5 MPa). After the effects of bottom cycle parameters on cycle performance and recovery efficiency of exhaust gas energy were studied, the design philosophy of bottom cycle parameters was obtained. On this basis, the bottom cycle concept was applied on IC engine full load, and the improvement rates of IC engine performances were studied under various speeds.

Table 2
The basic parameters of target IC engine.

Item	Content
IC engine type	Inline four cylinders
Bore (mm)	86
Stroke (mm)	86
Displacement (L)	1.998
Compression ratio	9.8
Connecting rod length (mm)	149.5
Valve number of each cylinder	4
Ignition mode	1-3-4-2
Rated power (kW)	110
Max torque (N m)	192
Cooling mode	Water-cooled
Intake mode	Naturally aspirated

In order to provide boundary conditions for bottom cycle performance calculation, we carried out the steady-state heat balance experiment on the original gasoline engine. And the experimental data under the target operating condition point (speed = 3000 r/min, BMEP = 0.5 MPa) are given in Table 3. Then, the experiment data of original four-cylinder engine are converted into that of three-cylinder baseline engine, which are also listed in Table 3. In this study, the exhaust gas temperature at heat exchanger (boiler) outlet is assumed to be 140 °C.

3.2. Bottom cycle parameters setting up and performance analysis

It is generally accepted that to analyze the T–S diagram and H–S diagram of working medium is a good method to study the thermodynamic cycle. Based on programming and fitting on computer, the T–S diagram and H–S diagram of steam were acquired, as shown in Figs. 4 and 5, respectively. Firstly, the initial states of working medium and the constraint conditions of this bottom cycle are defined. Because the outlet temperature of IC engine cooling water is around 90 °C, and the mass flow rate of cooling water is much larger than that of working medium water, the preheated temperature of working medium water is assumed to be 80 °C. The steam pressure after expansion is set to 0.15 MPa, which is referenced to the exhaust gas pressure of firing cylinder. In order to ensure that the steam after expansion does not contain liquid water, steam after expansion should be dry saturated steam or superheated steam (That is to say, steam mass fraction after expansion should be 100%). Consequently, the steam state after expansion is determined and listed in Table 4. At the same time, the corresponding isentropic expansion line which passes through the defined steam state point (saturated steam after expansion) is determined, as shown in the red line in Fig. 4. It means that the steam state before expansion should be on the isentropic line or at the right side of this isentropic line so as to ensure the steam mass fraction after expansion is 100%. In this study, we only discussed the former. In other words, steam state after expansion is assumed to be located at the saturated steam state point (As a result, steam state before expansion is on the isentropic line). As can be seen from the figure, there are many groups of cycle modes due to various kinds of steam pressure before expansion. Different cycle requires different steam pressure, and different steam pressure corresponds to different steam temperature since the isentropic expansion line is fixed. Therefore, the required steam temperature can be acquired from the intersection points of the superheated steam lines (isobaric lines) and isentropic expansion line. For example, when the steam pressure is set to 3.07 MPa, the required steam temperature should be 500 °C (superheated state).

Table 3
Performance experimental data of target IC engine.

Items	Original four-cylinder engine	Three-cylinder baseline engine
Speed (r/min)	3000	3000
BMEP (MPa)	0.5	0.5
Relative air-fuel ratio	1.008	1.008
Exhaust gas mass flow rate (g/s)	29.31	21.98
Power (kW)	26.35	19.76
Exhaust gas temperature at superheater inlet (°C)	670	670
Exhaust gas energy flow rate (kW)	23.3	17.5
Thermal efficiency (%)	32.6	32.6
Percentage of exhaust gas energy in total fuel energy (%)	28.8	28.8

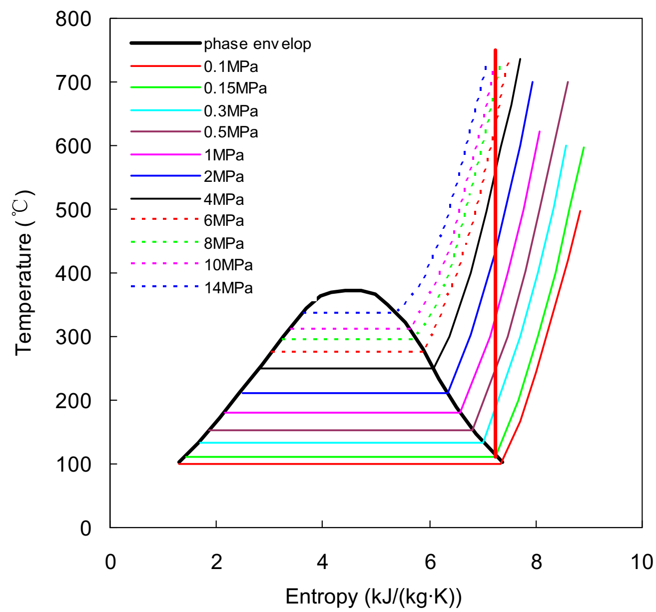

Fig. 4. The T–S diagram of water and steam for bottom cycle calculation.

Next, the water and steam H–S diagram (Fig. 5) is also analyzed for bottom cycle calculation and parameters setting up. The capacity for doing work of steam power cycle and even the recovery potential of exhaust gas energy can be concluded by analyzing this figure. Compared to the T–S diagram, H–S diagram is more convenient to study steam expansion work for it gives the specific enthalpy of each steam state point. As can be seen from the figure, the “sensitive area” which has higher specific enthalpy focuses on the upper right part of the H–S figure. The higher the steam pressure is, the more the steam expansion work will be. However, when the steam pressure increases to a certain level, the increase rate of specific enthalpy will become smaller. Instead, increasing steam “entropy” makes the steam state point shift to the right part of the H–S figure. By this means, the steam can obtain larger capacity for doing work since it has higher specific enthalpy.

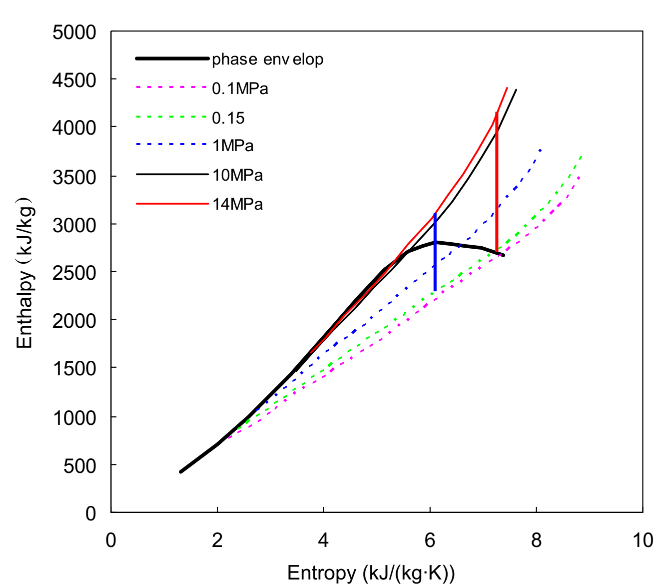

Fig. 5. The H–S diagram of water and steam for bottom cycle calculation.

Table 4
Initial conditions of the bottom cycle.

Item	Content
Initial water pressure (MPa)	0.1
Preheated water temperature (°C)	80
Steam pressure after expansion (MPa)	0.15
Steam mass fraction after expansion (%)	100
Steam temperature after expansion (°C)	111.37
Isentropic efficiency of pump	0.85
Isentropic efficiency of steam expansion	0.75

Actually, as the steam state after expansion is determined, the corresponding isentropic line is also determined, as shown in the red line in Fig. 5. Hence, the main way to improve steam specific enthalpy is to increase the steam pressure. That is to say, higher steam pressure results in higher steam expansion work for unit mass steam in each cycle.

In order to acquire the recovery potential of exhaust gas energy of this bottom cycle, the cycle processes were calculated by using computer program based on the boundary conditions given above. For the sake of simplifying the calculating processes, the pressure drop caused by friction loss and other irreversible loss along the pipe was ignored. After the thermodynamic parameters of each state point and the entire thermodynamic performances of the bottom cycle were calculated, the recovery potential of exhaust gas energy and improvement rate of IC engine thermal efficiency were drawn. The results and analysis were given in the following section.

4. Calculation results and analysis

4.1. Cycle processes calculation and analysis

As the steam state after expansion is fixed, the effects of steam parameters before expansion on the bottom cycle performances, especially the exhaust gas energy recovery efficiency are discussed in the next. For the main parameters of superheated steam are pressure and temperature, firstly, the relationship between steam pressure and steam temperature is analyzed. To realize the steam power cycle under the target operating condition and boundary conditions determined above, the relationship between the steam pressure and steam temperature is given in Fig. 6. For example, if the steam temperature at superheater outlet is designed to 500 °C, the corresponding steam pressure should be set to 3.07 MPa, as shown in the blue line. Then, the design condition point (steam pressure = 3.07 MPa, outlet steam temperature = 500 °C) is conducted for pinch point analysis, as shown in the H–T diagram of Fig. 7(a). As can be seen from this figure, the pinch point is located

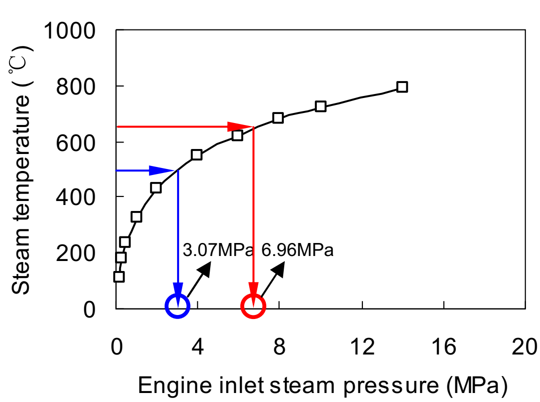

Fig. 6. The required heating temperature vs. steam pressure.

at the saturated liquid state of working medium. And the heat transfer parameters of this case (case 1) are listed in Table 5. In fact, Fig. 6 is transformed from Fig. 4. As what mentioned before, higher steam expansion work requires higher steam pressure. Now, the question is that what's the maximum pressure can be set under the fixed condition? As shown in Fig. 6, higher steam pressure corresponds to higher steam temperature. However, the steam heating temperature is limited by lots of parameters, including the temperature and mass flow rate of IC engine exhaust gas, heat exchanger efficiencies and mass flow rate of working medium water, etc. Under the target IC engine operating condition (speed = 3000 rpm, BMEP = 0.5 MPa), the exhaust gas temperature is fixed to 670 °C. Consequently, the highest steam temperature should be lower than exhaust gas temperature. In this study, the maximum outlet temperature of steam is assumed to be 650 °C. At the same time, the highest steam pressure can be determined to 6.96 MPa according to Fig. 6. Then, the heat transfer parameters of this condition are also analyzed by using pinch point approach, as shown in Fig. 7(b). In this case (case 2), the pinch point is located at the outlet of superheater with a temperature difference of 20 °C. And the heat transfer parameters of this case (case 2) are listed in Table 6.

The relationship between bottom cycle thermal efficiency and engine inlet steam pressure is shown in Fig. 8. As shown in the figure, the higher the steam pressure is, the higher the cycle efficiency will be. At the design condition (steam pressure is 3.07 MPa, and steam temperature is 500 °C), the cycle efficiency can reach 18.2%. Because a large part of exhaust gas energy recovered is in the form of steam latent heat and cannot be released through steam expansion, the steam power cycle efficiency is limited. When the steam pressure is at the highest level of 6.96 MPa (the corresponding steam temperature is 650 °C), the cycle efficiency can reach the maximum value of 23.3%. Combined with Figs. 6 and 8, some consequences can be concluded. On condition that steam state after expansion is fixed, higher steam pressure requires higher steam temperature, and both of them result in higher cycle efficiency. However, the highest steam temperature is limited by the temperature of IC engine exhaust gas. As a result, the highest steam pressure is also limited by the temperature of IC engine exhaust gas. Under the target IC engine operating condition (speed = 3000 r/min, BMEP = 0.5 MPa), the exhaust gas temperature is determined. Consequently, the highest steam temperature and the corresponding steam pressure are limited. Finally, the relationship between the maximum cycle efficiency and the temperature of IC engine exhaust gas is determined.

As the IC engine operating condition is fixed, the flow rate of exhaust gas available energy is also determined. Then, the mass flow rate of steam depends on steam heating temperature. In other words, for unit mass steam, the higher steam heating temperature requires the more exhaust gas heat. As a result, the higher the steam heating temperature is, the less the generated steam will be since the exhaust gas available energy flow rate is fixed. According to the analysis of Fig. 6, different steam temperature corresponds to different steam pressure. On this basis, the relationship between steam mass flow rate and engine inlet steam pressure is also established, and it is given in Fig. 9. As can be seen from the figure, the steam mass flow rate monotonically decreases with the increasing of steam pressure. This is because steam temperature increases as steam pressure increases. At the design condition (steam pressure is 3.07 MPa, and steam temperature is 500 °C), the corresponding steam mass flow rate is 4.37 g/s.

Fig. 10 shows the relationship between the output power of bottom cycle and engine inlet steam pressure. As shown in the figure, bottom cycle output power increases with the steam pressure, but the increase rate becomes gradually smaller. When the

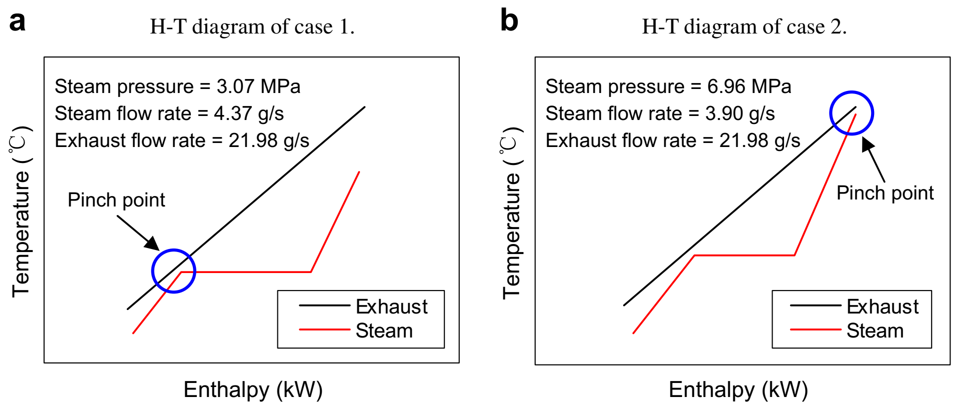

Fig. 7. H–T diagrams corresponding to two typical cases.

steam pressure is in the range of 0–4 MPa, the increase rate of bottom cycle output power is relatively high. However, when the steam pressure is greater than 8 MPa, the increase rate of bottom cycle output power is quite low. Therefore, from the viewpoint of exhaust gas energy recovery, when the steam pressure is larger than a certain range, there is no practical significance to improve steam pressure although cycle efficiency increases. This consequence can be also obtained by analyzing Fig. 5 (water and steam H–S figure). At the design condition (steam pressure is 3.07 MPa, and mass flow rate is 4.37 g/s), the bottom cycle output power is 2.5 kW. Meanwhile, when the steam pressure reaches 6.96 MPa, the bottom cycle output power comes up to the maximum value of 3.2 kW, which is much smaller than single firing cylinder power of 6.59 kW under the target operating condition. Consequently, if this bottom cycle concept is applied on engine part load, the total output power should decrease compared to the original four-cylinder engine.

Fig. 11 shows the recovery efficiency of exhaust gas energy using this steam power cycle. As the flow rate of exhaust gas energy is fixed under the target operating condition, the recovery efficiency of exhaust gas energy is determined by bottom cycle output power. This is the reason why Fig. 11 is very similar to Fig. 10. Also, recovery efficiency of exhaust gas energy increases with the rising of steam pressure. In the case of design condition (steam pressure is 3.07 MPa, and mass flow rate is 4.37 g/s), the recovery efficiency of exhaust gas energy is 14.3%. When steam pressure is at the maximum value of 6.96 MPa under target operating condition, the recovery efficiency of exhaust gas energy comes up to the maximum value of 18.3%. In fact, the influence factor of recovery efficiency of exhaust gas energy is not only the steam pressure, but also the steam temperature. As higher steam pressure requires higher steam temperature, finally both of them result in higher recovery efficiency of exhaust gas energy.

The ultimate purpose of this bottom cycle is to improve the IC engine thermal efficiency. Fig. 12 gives the improvement rate of IC engine thermal efficiency on the basis of three-cylinder baseline engine. Actually, the improvement rate of IC engine thermal efficiency depends on the recovered exhaust gas energy (steam expansion power). As can be seen from Fig. 12, the improvement rate of IC engine thermal efficiency also increases with the rising of steam pressure. At the design condition (steam pressure is 3.07 MPa, and mass flow rate is 4.37 g/s), the improvement rate of IC engine thermal efficiency is 4.1%. At the maximum steam pressure condition (steam pressure is 6.96 MPa) under target operating condition, the improvement rate of IC engine thermal efficiency can reach the maximum value of 5.3%. At that time, the total thermal efficiency of IC engine can come up to 37.9%. Consequently, there is a large energy saving potential when this bottom cycle is applied on IC engine for recovering exhaust gas energy.

4.2. Sizing the steam expansion cylinder

In this steam power cycle, the steam expansion ratio is a very important parameter to determine the geometry parameters of steam expansion cylinder. Firstly, the definition of required steam expansion ratio is given as

$$\gamma = \frac{\upsilon_{out}}{\upsilon_{in}} \tag{9}$$

where, γ is the required steam expansion ratio; υ_{in} is the steam specific volume before expansion while υ_{out} is the steam specific volume after expansion. Fig. 13 shows the required steam expansion ratio under different engine inlet steam pressure. As shown in Fig. 13, higher steam pressure corresponds to higher required steam expansion ratio. In the actual cycle process, steam expansion ratio is an important factor which determines the steam expansion process. On the one hand, if the actual expansion ratio is lower than the required steam expansion ratio, the steam will not fully expand. Under that circumstance, the steam pressure at cylinder BDC will be larger than 0.15 MPa, so part of steam pressure energy will be

Table 5
Heat transfer parameters of case 1.

Parameter	Content
Inlet exhaust gas temperature	670 °C
Mass flow rate of exhaust gas	21.98 g/s
Steam pressure	3.07 MPa
Mass flow rate of steam	4.37 g/s
Outlet steam temperature	500 °C
Efficiency of boiler	0.88
Efficiency of superheater	0.61

Table 6
Heat transfer parameters of case 2.

Parameter	Content
Inlet exhaust gas temperature	670 °C
Mass flow rate of exhaust gas	21.98 g/s
Steam pressure	6.96 MPa
Mass flow rate of steam	3.90 g/s
Outlet steam temperature	650 °C
Efficiency of boiler	0.87
Efficiency of superheater	0.95

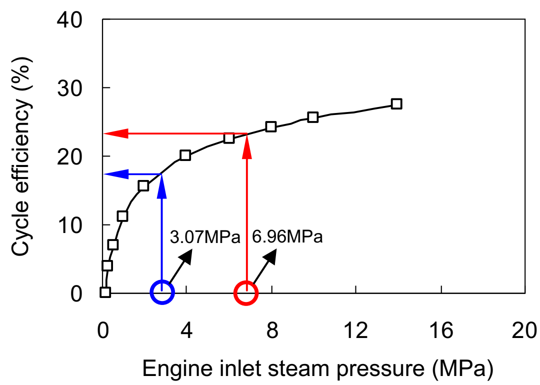

Fig. 8. Cycle efficiency of steam power cycle vs. steam pressure.

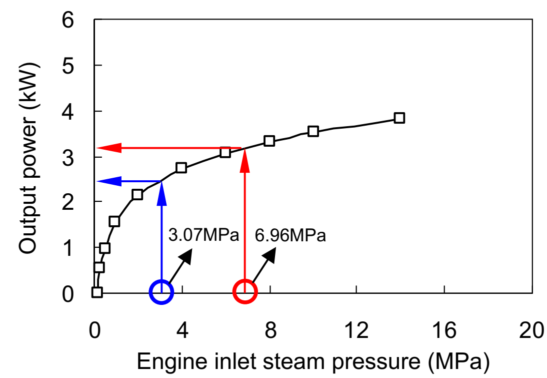

Fig. 10. The output power of this bottom cycle.

wasted. On the other hand, if the actual expansion ratio is larger than the required steam expansion ratio, the steam will excessively expand. Under that circumstance, the steam mass fraction may be lower than 100% at cylinder BDC. At the design condition (steam pressure is 3.07 MPa, and mass flow rate is 4.37 g/s) the required steam expansion ratio is 10.22, which is a bit larger than that of original IC engine. Next, the actual expansion ratio (or effective expansion ratio) of steam expansion cylinder should be modified. In fact, it can be adjusted through changing the timing of intake valve, and this part content will be discussed in the next section. When the steam pressure is at the maximum value of 6.96 MPa, the required steam expansion ratio reaches 19.4. Under that circumstance, the actual expansion ratio of steam expansion cylinder should be redesigned, and it seems easier to realize on the diesel engine.

Then the question is that how to judge whether the steam volume flow rate matches the steam expansion cylinder under various engine speeds. Firstly, the formulas for two kinds of volume flow rate of steam are given as follows:

$$\dot{V}_{\mathrm{inj}} = \frac{n}{60}\cdot V_1 \tag{10}$$

$$\dot{V}_{\mathrm{st}} = \dot{m}_{\mathrm{st}}\cdot v_{\mathrm{in}} \tag{11}$$

where, $\dot{V}_{\mathrm{inj}}$ is the maximum volume of steam can be injected into cylinder in per second; n is the IC engine speed; V_1 is the cylinder volume at the intake valve close angle, as shown in the Fig. 3(a); $\dot{V}_{\mathrm{st}}$ is the volume flow rate of steam generated by IC engine exhaust gas. If $\dot{V}_{\mathrm{st}}$ is less than $\dot{V}_{\mathrm{inj}}$, it will means that the steam generated by IC engine exhaust gas is not enough under this operating condition and all the steam can be used. Conversely, it will imply that the steam expansion cylinder is too little and part of steam cannot be used. As actual expansion ratio of the engine without VVT (variable valve timing) cannot be changed under different speed, this bottom cycle concept can only work well under fixed operating condition with fixed steam parameters (The ideal case is that $\dot{V}_{\mathrm{st}}$ equals $\dot{V}_{\mathrm{inj}}$. Under that circumstance, the bottom cycle has the optimum performance).

Then, the parameters of steam expansion cylinder are sized. In order to design the actual expansion ratio of steam expansion cylinder, the relationship between the cylinder volume and crank angle is given as

$$V = \frac{\pi D^2}{4}\cdot\frac{S}{2}\left[\left(1+\frac{1}{\lambda}\right)-\cos\left(\frac{\pi}{180}\cdot\phi\right) - \frac{1}{\lambda}\sqrt{1-\lambda^2\cdot\sin^2\left(\frac{\pi}{180}\cdot\phi\right)}\right]+V_0 \tag{12}$$

where, D is the cylinder bore; S is the cylinder stock; λ is the ratio of connecting rod and crank; V_0 is the redesigned cylinder clearance volume; ϕ is the crank angle.

The actual expansion ratio of steam expansion cylinder can be calculated according to the following formula:

$$\varepsilon = \frac{V_2}{V_1} = \frac{V_{\mathrm{dis}}+V_0}{V_1} \tag{13}$$

where, ε is the actual expansion ratio of steam expansion cylinder; V_2 is the maximum cylinder volume (piston is at the BDC); V_{dis} is the displacement of steam expansion cylinder, which is 0.5 L.

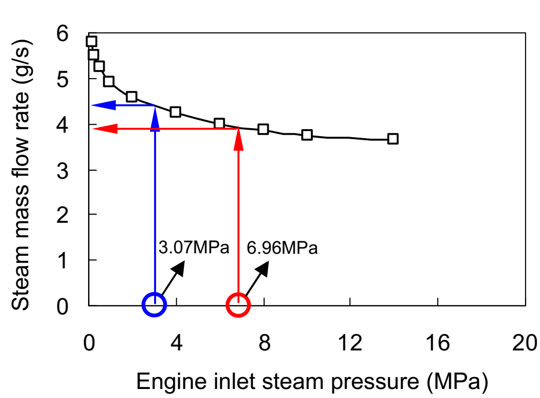

Fig. 9. Steam mass flow rate could be supplied by exhaust gas energy.

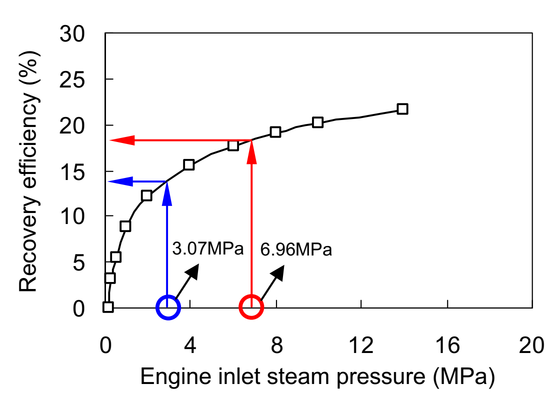

Fig. 11. Recovery efficiency of exhaust gas energy vs. steam pressure.

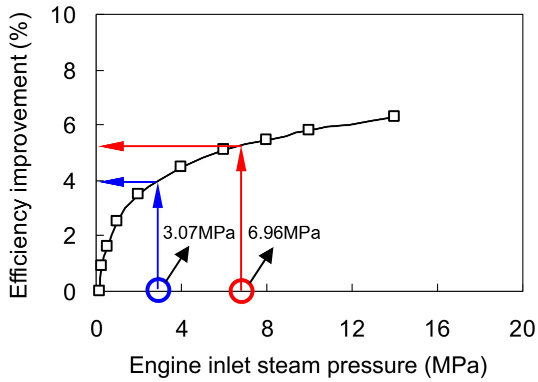

Fig. 12. The improvement rate of IC engine thermal efficiency vs. steam pressure.

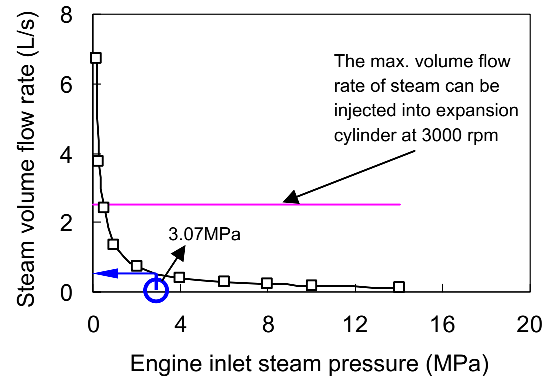

Fig. 14. Steam volume flow rate vs. steam pressure.

As mentioned above, at the design condition the required steam expansion ratio is 10.22. Therefore, the actual expansion ratio of steam expansion cylinder is designed to 10.22 under the target IC engine operating condition. In this study, the intake valve close angle is assumed to be 30°CA, thus the initial cylinder volume V_1 is 0.0496 L, which can be acquired according to Formula (12) and Formula (13). As a result, $\dot{V}_{inj}$ can be calculated according to Formula (10), and it is 2.48 L.

Fig. 14 shows the volume flow rate of steam could be supplied by exhaust heat energy under the target IC engine operating condition (as shown in the black line). As shown in Fig. 14, the volume flow rate of steam decreases fast when the engine inlet steam pressure is lower than 1 MPa. And it is nearly invariant when the steam pressure is greater than 4 MPa. Under the target IC engine operating condition (speed = 3000 r/min, BMEP = 0.5 MPa), the maximum volume flow rate of steam can be injected into steam expansion cylinder is 2.48 L, and it is also plotted in Fig. 14 (as shown in the red line). However, the volume flow rate of steam generated by IC engine exhaust gas is 0.5 L. Thus, the volume flow rate of steam is not enough under this operating condition. This is because the exhaust gas energy is too low to generate the enough steam for steam expansion cylinder under this operating condition. Consequently, this bottom cycle concept may work better under higher or even full load due to more exhaust gas energy.

4.3. IC engine performance improvement

As can be known from the above analysis, on condition that steam state after expansion is fixed, higher inlet steam pressure corresponds to higher steam temperature, both of which result in higher cycle efficiency and recovery efficiency of exhaust gas energy. According to this principle, the recovery potential of exhaust gas energy and the improvement rate of IC engine thermal efficiency were studied under IC engine full load. Firstly, the test data of IC engine exhaust gas under full load are listed in Table 7, and the calculation results of bottom cycle output power and the improvement rate of IC engine thermal efficiency are given in Figs. 15 and 16, respectively.

As can be seen in Fig. 15, bottom cycle output power increases with the IC engine speed. The main reason is that exhaust gas energy increases with the speed. At the speed of 1000 r/min, bottom cycle output power is only 0.77 kW. While at the speed of 6000 r/min, bottom cycle output power reaches 19.2 kW, which is still lower than the power of each firing cylinder. Compared to the original four-cylinder engine, the total output power of IC engine with steam expansion cycle still decreases even under full load. In the meantime, the improvement rate of IC engine thermal efficiency also increases with the speed, as shown in Fig. 16. Two fundamental factors can account for this phenomenon: firstly, the percentage of exhaust gas energy in total fuel energy increases with the rising of IC engine speed [7]; secondly, as the IC engine speed rises, exhaust gas temperature also increases, and it results in the improving of cycle efficiency. From the viewpoint of exergy analysis, it's not difficult to know that the percentage of exhaust gas exergy is mainly determined by exhaust gas temperature. At the speed of 6000 r/min, the improvement rate of IC engine thermal efficiency comes up to the maximum value of 6.3%. Moreover, at the speed of 4000 r/min, the total thermal efficiency of IC engine reaches the maximum of 39.0%. At that time, the IC engine thermal efficiency is improved by 5.9%. Consequently, IC engine energy utilization efficiency is greatly improved and the goal of energy saving and emission reduction can be realized by using this bottom cycle to recover IC engine exhaust gas energy.

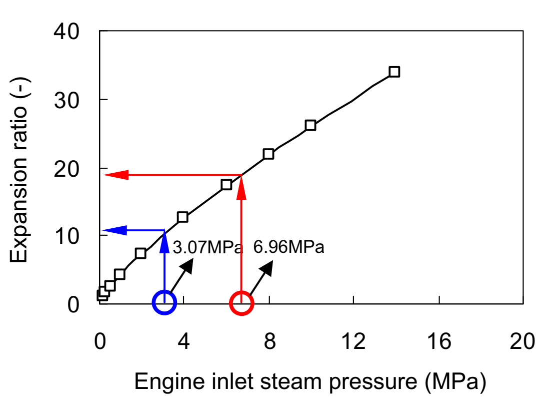

Fig. 13. The required steam expansion ratio vs. steam pressure.

Table 7
IC engine performance experimental data under full load (original four-cylinder engine).

Speed (r/min)	Exhaust gas temperature (°C)	Exhaust gas mass flow rate (g/s)	IC engine power (kW)
1000	502.62	14.30	14.41
2000	596.33	33.10	32.98
3000	691.98	50.39	50.54
4000	729.64	75.99	75.03
5000	774.65	100.49	94.50
6000	800.55	119.43	105.09

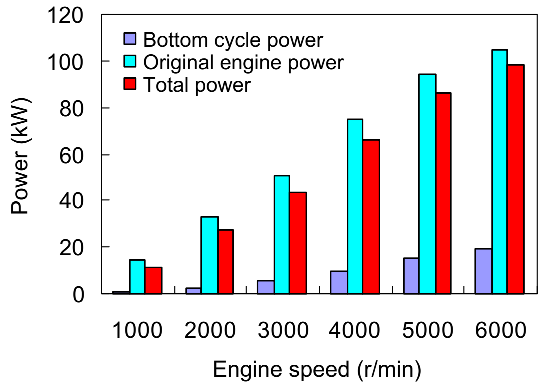

Fig. 15. Bottom cycle output power vs. IC engine power under full load.

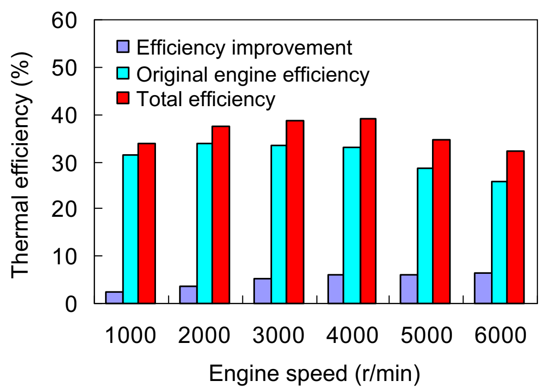

Fig. 16. Improvement rate of thermal efficiency vs. original engine efficiency.

5. Conclusions

In this paper, a novel bottom cycle concept used for recovering IC engine exhaust gas energy based on open steam expansion cycle was proposed. Compared to other recovery means, this bottom cycle concept is easier to realize on IC engine since the cost is low while recovery efficiency is relatively high. Through calculating and analyzing the bottom cycle processes, some conclusions are drawn.

(1) This novel concept can recover both the exhaust gas energy and cooling water energy by the combined steam power cycle. As the steam expansion cylinder has higher expansion ratio, steam can expand more fully in cylinder than in turbine. Therefore, it can achieve higher waste heat recovery efficiency than other Rankine steam cycles using turbine for steam expansion. Also, because the steam expansion cylinder is coupled on the crankshaft, the recovered exhaust gas energy can be directly used to drive the automobile or other loads with high transmission efficiency.

(2) The recovery efficiency of exhaust gas energy is influenced by working medium latent heat. Because steam has a large latent heat, the cycle efficiency is limited since a large part of exhaust gas energy recovered is in the form of steam latent heat which cannot be outputted by expanding. At the same time, because the steam with higher temperature can output more expansion work on condition that steam latent heat is fixed, the higher exhaust gas temperature results in greater energy recovery efficiency. As a result, the energy recovery efficiency of this bottom cycle is limited by IC engine exhaust gas temperature ultimately. In addition, this bottom cycle concept can work better under higher load and higher speed. On the other hand, the steam expansion ratio is restricted by the expansion cylinder. Normally, the expansion ratio of cylinder cannot be changed during IC engine operating process, unless the valve system is equipped with VVT. From this viewpoint, this bottom cycle seems to work better at fixed working condition under full load, especially in stationary applications, e.g. Genset.

(3) Under the fixed operating condition of IC engine part load, the maximum recovery efficiency of exhaust gas energy is 18.3%; the maximum steam expansion power is 3.2 kW; and IC engine thermal efficiency can be increased by 5.3%. All those can demonstrate this concept for exhaust gas energy recovery has great energy saving potential and good application prospect.

(4) Under IC engine full load, the higher recovery efficiency of exhaust gas energy appears at higher speed since IC engine has higher exhaust gas temperature, and the maximum improvement rate of IC engine thermal efficiency can reach 6.3% at 6000 r/min. However, the bottom cycle output power is still lower than the power of firing cylinder even under full load. This is because the exhaust gas energy is not high enough to generate the enough steam. For this reason, when this bottom cycle concept is applied on six-cylinder or even eight-cylinder engines, it may work better without the decreasing of total power compared to original engine.

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Nomenclature

η	efficiency [%]
h	specific enthalpy [kJ/kg]
T_0	environment temperature [K]
c_P	specific heat at constant pressure [kJ/(kg k)]
P	power [kW]
$\dot{m}$	mass flow rate [kg/s]
T	temperature [K], [°C]
$\dot{\Phi}$	heat flux [kJ/s]
H_u	fuel low heat value [kJ/kg]
γ	the required steam expansion ratio
υ	specific volume [m^3/kg]
$\dot{V}$	volume flow rate [m^3/s]
n	IC engine speed [r/min]
V	volume [m^3]
D	cylinder bore [mm]
S	cylinder stock [mm]
λ	ratio of connecting rod and crank
V_0	the redesigned cylinder clearance volume [m^3]
ϕ	crank angle [CA]
ε	actual expansion ratio of steam expansion cylinder

Subscripts

pum	pump
st	steam
cyl	cylinder
bot	bottom cycle
h	heat transfer

ex exhaust gas
re recovery
im improvement rate
fue fuel
in inlet
out outlet
inj injection
dis displacement

Abbreviation
ORC organic Rankine cycle
BDC bottom dead center
TDC top dead center
IMEP indicated mean effective pressure
BMEP brake mean effective pressure
PMEP pumping mean effective pressure
NMEP net mean effective pressure
CA crank angle
VVT variable valve timing

References

[1] Gopal NK, Subbarao R, Pandiyarajan V, Velraj R. Thermodynamic analysis of a diesel engine integrated with a PCM based energy storage system. International Journal of Thermodynamics 2010;13(1):15–21.
[2] Srinivasan KK, Mago PJ, Krishnan SR. Analysis of exhaust waste heat recovery from a dual fuel low temperature combustion engine using an Organic Rankine Cycle. Energy 2010;35(6):2387–99.
[3] Kalian N, Zhao H, Yang C. Effects of spark-assistance on controlled auto ignition combustion at different injection timings in a multicylinder direct injection gasoline engine. International Journal of Engine Research 2009; 10(3):133–48.
[4] Sudheesh K, Mallokarjuna JM. Diethyl ether as an ignition improver for biogas homogeneous charge compression ignition (HCCI) operation – an experimental investigation. Energy 2010;35(9):3614–22.
[5] Yuksel F, Ceviz MA. Thermal balance of a four stroke SI engine operating on hydrogen as a supplementary fuel. Energy 2003;28(11):1069–80.
[6] Taymaz I. An experimental study of energy balance in low heat rejection diesel engine. Energy 2006;31(2):364–71.
[7] He MG, Zhang XX, Zeng K, Gao K. A combined thermodynamic cycle used for waste heat recovery of internal combustion engine. Energy 2011;36(12): 6821–9.
[8] Hung TC, Wang SK, Kuo CH, Pei BS, Tsai KF. A study of organic working fluids on system efficiency of an ORC using low grade energy sources. Energy 2010; 35(3):1403–11.
[9] Roy JP, Mishra MK, Misra A. Parametric optimization and performance analysis of a waste heat recovery system using Organic Rankine Cycle. Energy 2010;35(12):5049–62.
[10] Guo J, Xu M, Cheng L. Thermodynamic analysis of waste heat power generation system. Energy 2010;35(7):2824–35.
[11] Prater DM. A dual cycle reciprocating engine. SAE technical paper 2000-01-3070.
[12] Conklin JC, Szybist JP. A highly efficient six-stroke internal combustion engine cycle with water injection for in-cylinder exhaust heat recovery. Energy 2010; 35(4):1658–64.
[13] Kapich D. Turbo-hydraulic engine exhaust power recovery system. SAE Technical Paper 2002-01-2731.
[14] Hsiao YY, Chang WC, Chen SL. A mathematic model of thermoelectric module with applications on waste heat recovery from automobile engine. Energy 2010;35(3):1447–54.
[15] Shen WD, Jiang ZM, Tong JG. Engineering thermodynamics. 3rd ed. Beijing: Higher Education Press; 2001 [in Chinese].
[16] Wang EH, Zhang HG, Fan BY, Ouyang MG, Zhao Y, Mu QH. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. Energy 2011;36(5):3406–18.
[17] Liu BT, Chein KH, Wang CC. Effect of working fluids on Organic Rankine Cycle for waste heat recovery. Energy 2004;29(8):1207–17.
[18] Liu JP, Fu JQ, Feng K, Wang SQ, Zhao ZC. Characteristics of engine exhaust gas energy flow. Journal of Central South University (Science and Technology) 2011;42(11):3370–6 [in Chinese].